State-of-Art Survey

DEVELOPMENT OF A SOLAR-POWERED RESIDENTIAL AIR CONDITIONER

Contract NAS8-30758

74-10996(3)

January 13, 1975

Prepared for

George C. Marshall Space Flight Center
National Aeronautics and Space Administration
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Huntsville, Alabama 35812

AIRESEARCH MANUFACTURING COMPANY
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GARRETT

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SECTION 1

INTRODUCTION AND SUMMARY

INTRODUCTION

An extensive review of the literature was conducted in partial fulfillment of Task 1 of Contract NAS 8-30758. This review was concerned with (1) the characterization of systems and equipment that could be applicable to the development of solar-powered air conditioners based on the Rankine-cycle approach, and (2) the establishment of baseline data defining the performance, physical characteristics, and cost of systems using the LiBr/H_2O absorption cycle. The baseline data are to be used later in the study for comparision with an optimized Rankine-driven vapor compression system. This report summarizes the information gathered to date and discusses published data in terms of the objectives of the present study. Since significant study and development work is in progress, the state-of-the-art survey will be continued through the contract period and the results of current programs reviewed as information becomes available. Additional information collected during the state-of-the-art review will be reported in the monthly progress reports.

A brief summary of the findings is given below. Summaries and discussions of the most pertinent data are presented in Sections 2 and 3, and references are listed in Section 4.

SUMMARY

Air conditioners specifically designed for operation using solar thermal energy are not presently manufactured commercially. However, the technology is available for the development and production of such equipment, and several organizations are engaged in the development of prototype components and systems for this application. Most air conditioners installed in experimental solar houses have been conventional electric-driven vapor compression units or absorption-type coolers. The latter are obsolete Arkla LiBr/H_2O absorption units modified for this specific application; the modification involves replacing the gas-fired generator with a water-fired unit to accommodate the solar collector subsystem interface. Currently under NSF sponsorship, Arkla is further modifying an existing water chiller design to incorporate a recirculation pump and an evaporatively cooled absorber. The anticipated performance of the unit is listed in Table 1-1 at the operating conditions noted.

With respect to Rankine-driven refrigeration systems, the data listed in Table 1-2 characterize the performance of such systems using state-of-the-art compressor and turbine designs. These values agree fairly well with experimental data obtained on a Barber-Nichols system developed for installation and evaluation in the Honeywell mobile solar laboratory.

TABLE 1-1

ESTIMATED PERFORMANCE OF WATER-FIRED ABSORPTION AIR CONDITIONER

Cooling capacity	10.54 Kw (3 tons)
Hot water source temperature	363.7°K in/358.2°K out (195°F in/185°F out)
Chilled water temperature	285.9°K in/280.4°K out (55°F in/45°F out)
Evaporative heat rejection	298.7°K (78°F wb air in)
Water consumption	25.2 μm^3/sec (24 gal/hr)
Coefficient of performance	0.65
Electrical consumption	875 watts maximum

TABLE 1-1

TYPICAL STATE-OF-ART RANKINE AIR CONDITIONER PERFORMANCE DATA

Hot water inlet temperature	377.6°K (220°F)
Evaporator temperature	280.4°K (45°F)
Condensing temperature	305°K (90°F)
Rankine-cycle efficiency	0.1
Vapor-compression cycle COP	6.0
Overall COP	0.6

The conclusions reached by various investigators as to the economic applicability of solar-powered air conditioners are widely conflicting (References 1-1, 1-2, and 1-3). Overall system life-cycle cost and pay-back data vary by more than an order of magnitude. The discrepancies between the published data are due to the different assumptions basic to these studies. These assumptions are primarily related to (1) the projected cost of solar collectors, (2) the projected cost of fuel, and (3) the effectiveness of the heat-powered refrigeration system configuration and utilization. It is apparent from these studies

that the cost of the solar collector is a decisive factor in determining pay-back time. Since the size of the collector is directly related to the effectiveness of the refrigeration system, every effort should be made to develop a refrigeration machine designed for maximum efficiency and low fabrication cost. The technology for this is available.

As an example, the solar collector size necessary to power a Rankine-cycle air conditioner can be reduced by 30 percent as the efficiencies of the system compressor and turbine increase from 70 percent to 80 and 85 percent. In itself, this represents a significant factor in the performance of economic studies. It is an objective of the present study program to determine the potential of Rankine-powered air conditioning equipment optimized for operation at the low-temperature heat source attainable from flat-plate solar collectors. The future utilization of solar-powered air conditioners may not be determined from economic factors, but rather may be imposed by legislation aimed solely at resolving the nation's energy dependency.

SECTION 2

REVIEW OF ABSORPTION SYSTEM LITERATURE

GENERAL

An excellent treatment of absorption refrigeration systems is presented in Reference 2-1. At the time (1957), Servel manufactured lithium bromide/water ($LiBr/H_2O$) air conditioners for residential use. These were atmospheric-steam or water-cooled units. In an effort to eliminate the requirements for cooling towers as ultimate heat sinks for the $LiBr/H_2O$ absorber, much work was performed in the development of refrigerant-absorbent combinations with improved vapor pressure properties more suitable to the design of air-cooled systems. Such a program, sponsored by the American Gas Association (AGA), is summarized in Reference 2-2 (1968); data presented indicate that a $LiBr/LiSCN/H_2O$ fluid system has lower vapor pressure characteristics, and this offers potential for air-cooling the absorber without crystallization.

ABSORPTION SYSTEM AVAILABILITY

Absorption-type air conditioners have not penetrated the residential marketplace to any extent. Currently, Arkla does not produce any small-tonnage air conditioners. However, they do market gas-fired air-cooled water chillers in basic sizes of 10.5 kw (3 tons), 14.1 kw (4 tons), 17.6 kw (5 tons) and multiples thereof. Data typical of the performance of these water chillers are presented in Table 2-1 for the 14.1-kw (4-ton) unit. Coefficient of performance of 0.48 (excluding auxiliary electrical power) is specified; this corresponds to an ambient air temperature of 308.2°K (95°F) and a chilled water supply temperature of 280.4°K (45°F).

ABSORPTION SYSTEM EVALUATION WITH SOLAR COLLECTOR

Wisconsin University Tests

An early Arkla 10.5-kw (3-ton) $LiBr/H_2O$ absorption unit (Model DUCS-2) was tested in conjunction with a flat-plate solar collector at the University of Wisconsin in 1962 (Reference 2-4). In this installation, the generator (designed for steam heating) was heated with liquid water from the solar collector. The lower heat transfer coefficient obtained with water resulted in a reduction in capacity from 10.5 kw (3 tons) to 7.03 kw (2 tons). The absorption system provided cooling, with water temperatures at generator inlet as low as 355.4°K (180°F) and cooling water at 302.6°K (85°F) cooling water. A coefficient of performance between 0.4 and 0.6 was generally achieved on test.

Marshall Space Flight Center Tests

A water-cooled Arkla $LiBr/H_2O$ system was modified by replacing the gas-fired generator with a water-fired unit compatible for operation with a flat-plate solar collector. This modified unit was designed to provide 10.5 kw (3 tons) of refrigeration at a COP of 0.65 with water temeratures

TABLE 2-1

SPECIFICATION DATA FOR
ARKLA MODEL ACB 48-00 AIR-COOLED WATER CHILLER
(MODIFIED FROM REFERENCE 2-3)

Performance Data	
Capacity	14.1 kw (4 tons*)
Gas input	29.3 kw (100,000 Btu/hr)
Condenser air volume (approximate)	2.83, m^3/sec (6000 cfm)
Chilled Water Data	
Water quantity flow rate	10.1 μm^3/sec (9.6 gpm)
Maximum water flow rate	12.6 μm^3/sec (12 gpm)
Inlet water temperature	285.9°K (55.0°F)
Outlet water temperature	280.4°K (45.0°F)
Unit volume (approximate)	0.011 m^3 (3.0 gal)
Electrical	
Electrical	230v
Condenser fan, direct-drive motor (230 v, 60 cycle, 1 phase)	0.5 hp
Solution and chilled water pump motor (230 v, 60 cycle, 1 phase)	0.5 hp
Operating wattage draw (typical)	1000
Amperage draw	less than 12 amp
Fuse size: time delay (2)	15 amp
Dimensions	
Width	0.851 m (33-1/2 in.)
Depth	1.23 m (48-1/2 in.)
Height	1.07 m (42-1/4 in.)
Physical Data	
Operating weight (approximate)	340.2 kg (750 lb)
Shipping weight (approximate)	374.2 kg (825 lb)
Refrigerant type	717

*Refrigeration capacity shown is based on ambient temperature at 308.2°K (95°F), chilled water supply at 280.4°K (45°F), and at the flow rates specified.

of 372.0°K (210°F) at generator inlet, and 302.6°K (85°F) at absorber inlet. Such a unit was installed in the NASA Marshall Space Flight Center Solar House for experimental evaluation (Reference 2-5). Estimated performance for this version of the Arkla unit is presented in Reference 1-1. Figure 2-1 (taken from Reference 1-1) shows the sensitivity of the cooling capacity of the unit in terms of water temperature at the inlet of the generator and the absorber. For a 302.6°K (85°F) cooling water temperature (which represents a normal design value for cooling towers), the capacity of the unit will drop from 3 to 1.5 tons as the generator water inlet temperature drops from 372.0°K (210°F) to 355.4°K (180°F).

CURRENT DEVELOPMENT ACTIVITIES

At present, Arkla Industries (under NSF sponsorship) is engaged in a program aimed at the development of a laboratory prototype solar-powered $LiBr/H_2O$ absorption system specifically optimized for residential use. This unit will incorporate the following:

(a) Water-fired generator

(b) Liquid pump for circulation of the $LiBr/H_2O$ solution between the absorber and generator

(c) Direct evaporative cooling of the absorber and condenser

This laboratory prototype will be operated as a water chiller and will provide 10.5 kw (3 tons) of refrigeration under the following conditions (Reference 2-6).

(a)	Heat-source water temperature:	363.7°K in/358.2°K out (195°F in/185°F out)
(b)	Chilled water temperature:	285.9°K in/280.4°K out (55°F in/45°F out)
(c)	COP:	0.65
(d)	Evaporative heat rejection:	to 298.7°K (78°F) wt air in
(e)	Water consumption:	25.2 μm^3/sec
(f)	Electrical power consumption:	875 watts maximum

Although the size and cost of absorption refrigerators modified for solar heat utilization are not available at this time, it is believed (Reference 2-7) that the cost of such a machine will be about the same as that for currently marketed water chillers (see Table 2-2).

AIRESEARCH ABSORPTION SYSTEM INVESTIGATIONS

A computer program was developed by AiResearch to generate parametric $LiBr/H_2O$ absorption system performance data. The system modeled is illustrated

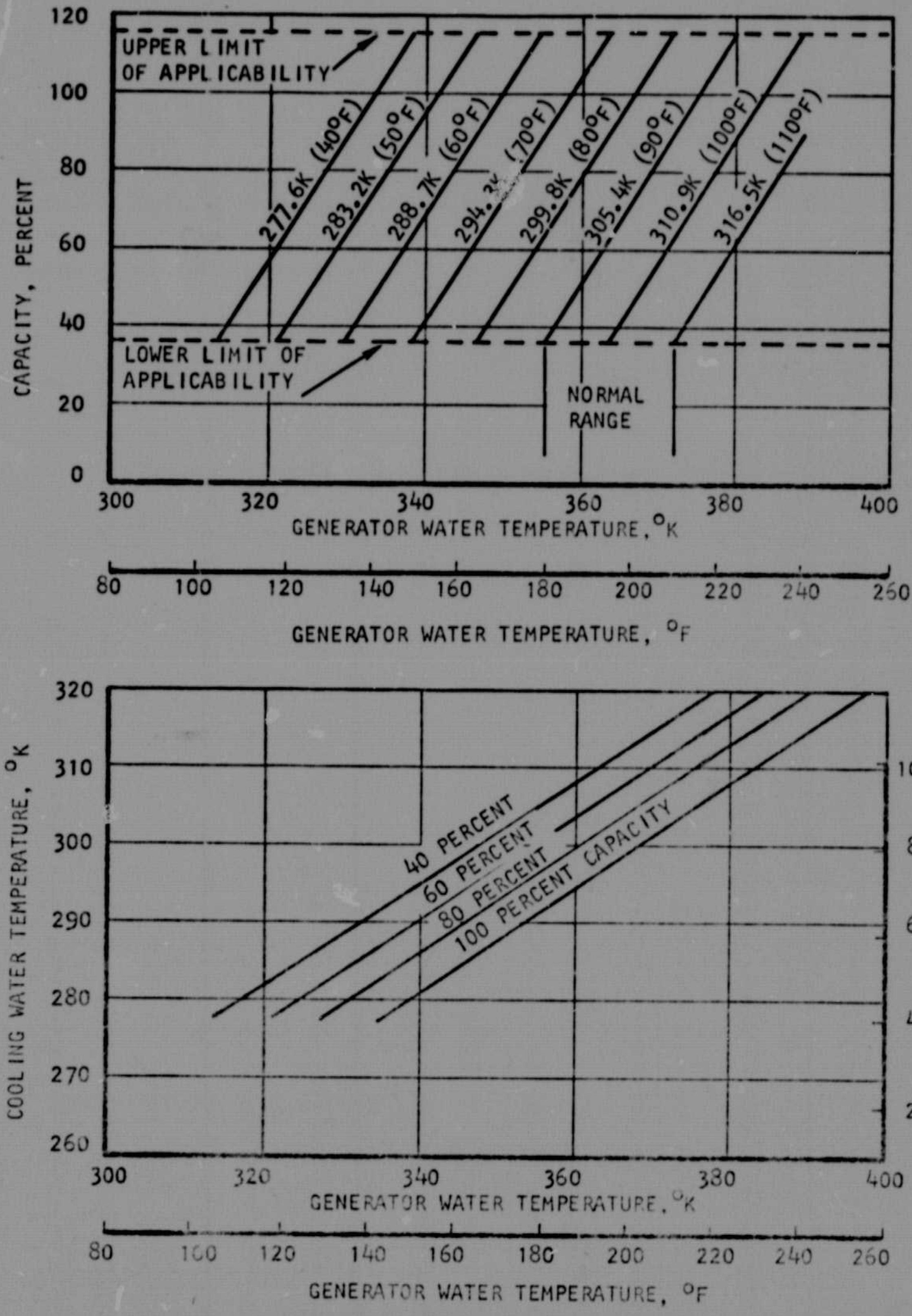

Figure 2-1. Performance of Modified Arkla LiBr/H_2O Refrigerator (From Reference 1-1)

TABLE 2-2

ARKLA LiBr/H_2O SYSTEM COST

Arkla Model No.	Capacity, kw (tons)	Distributor List Price, 1975 dollars
ACB 36-00	10.5 (3)	1265
ACB 48-00	14.1 (4)	1629
ACB 60-00	17.6 (5)	1845

in Figure 2-2. The program computes cycle COP for fixed values of the following system parameters.

(a) Generator temperature

(b) Absorber temperature

(c) Condenser temperature

(d) Evaporator temperature

(e) Evaporator load

(f) Recuperator effectiveness

(g) Absorber effectiveness

The computation procedure involves iterative material and heat balance calculations until the concentrations of the solution in the absorber and generator are such that the thermodynamic and mass transfer requirements established by the input parameters are satisfied. The pressure drop in each system element is taken into account and pump power is computed. This program was exercised over a range of conditions for fixed values of recuperator effectiveness (0.8) and absorber efficiency (0.6); these values appear realistic for the system considered. The data are presented in Figure 2-3 as cycle COP (evaporator load/generator heat input) plotted as a function of generator temperature and evaporator temperature. Each plot corresponds to a different condenser temperature.

Examination of the data indicates that for any combination of absorber-condenser-evaporator temperatures, the attainable COP remains about constant over a wide range of generator temperatures. However, as the generator temperature drops below a critical value, the design of a LiBr/H_2O absorption system becomes thermodynamically impossible. Since the computer program is a design program rather than a performance prediction program, the plots of Figure 2-3 cannot be used to predict off-design point performance because the concentration of the LiBr/H_2O solution varies for each condition represented by the

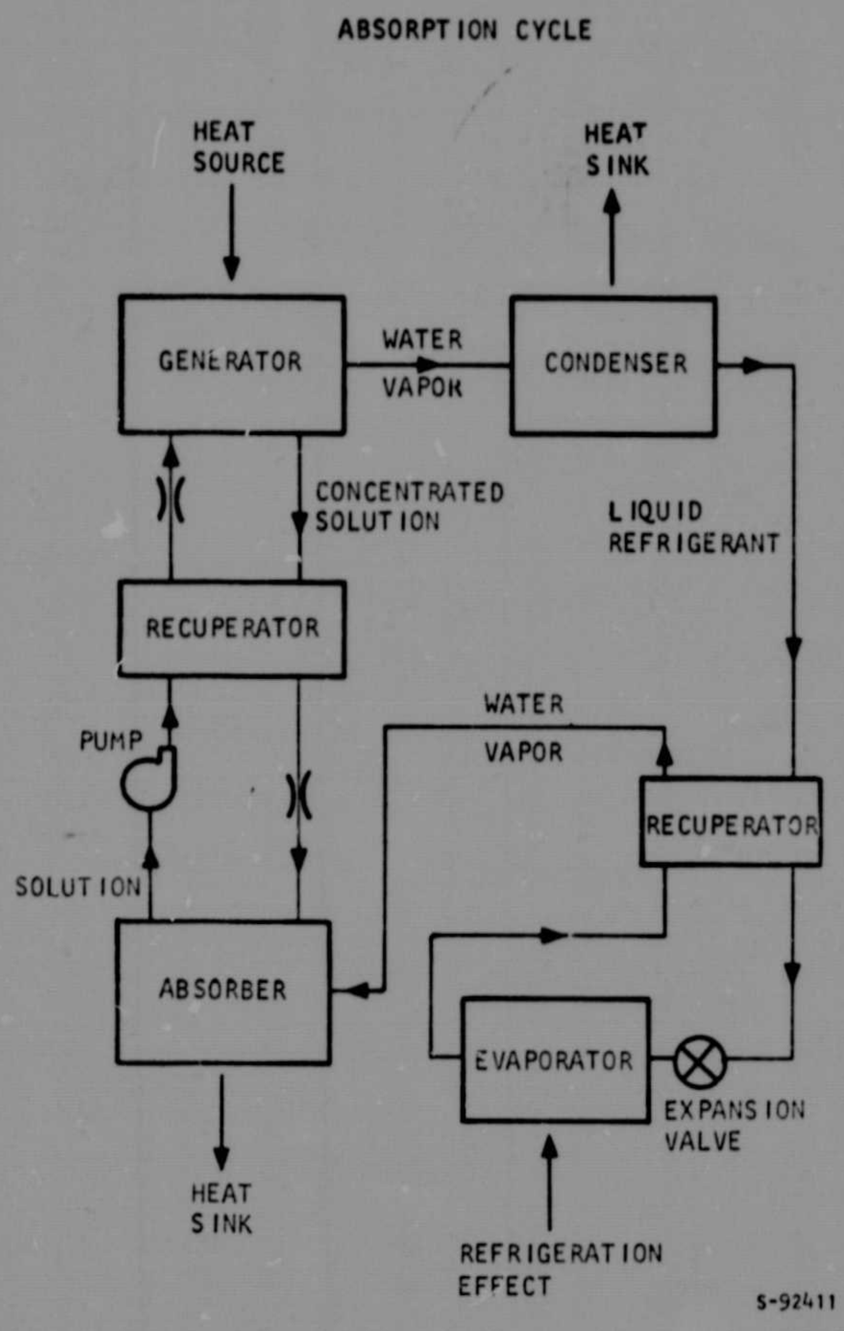

Figure 2-2. LiBr/H_2O Absorption System Schematic

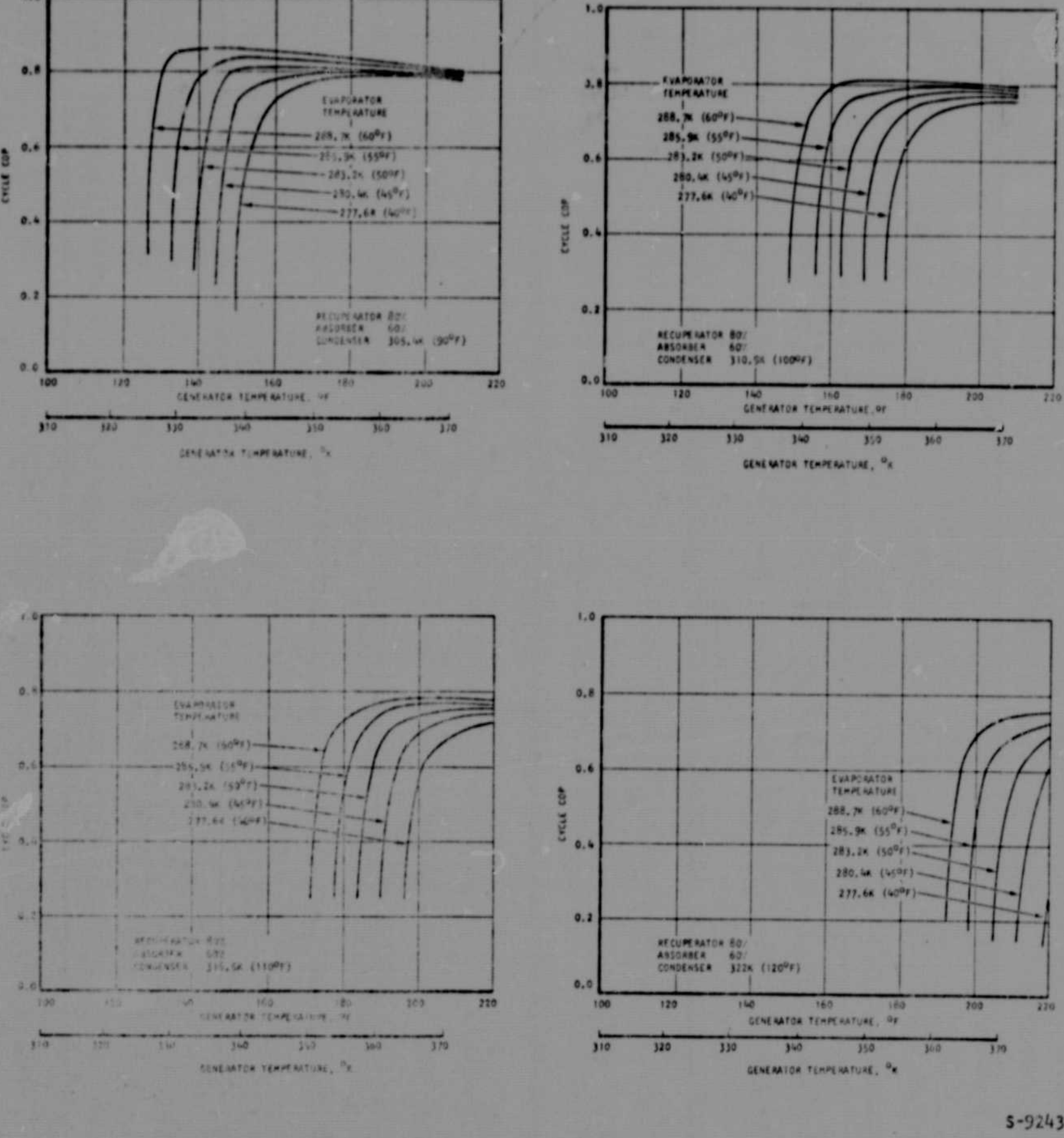

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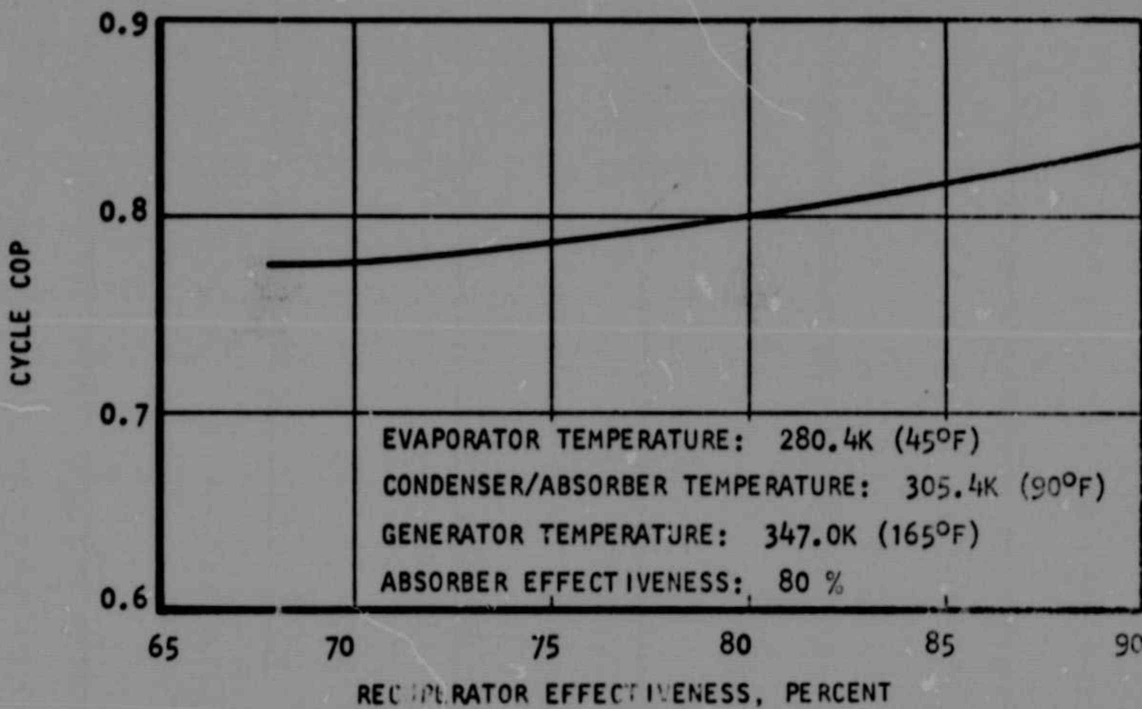
Figure 2-3. Effect of Condenser Temperature $LiBr/H_2O$ Refrigeration System

system temperatures. Referring to Figure 2-3, it is apparent that generator temperatures higher than 190°F are necessary to provide a heat sink temperature consistent with the requirements for an air-cooled air conditioner. It follows that the design of such a system will require relatively high effectiveness heat exchangers for operation. The effects of absorber and recuperator effectiveness on cycle COP for fixed values of system operating temperatures are shown in Figure 2-4. The plots show a relatively small change in COP as a function of these two parameters. The COP shown includes the effect of pressure drop on system performance. However, these data are somewhat optimistic because heat losses and gains from ambient are not considered in the calculation procedure. Such losses would easily result in a drop in COP as large as 0.1.



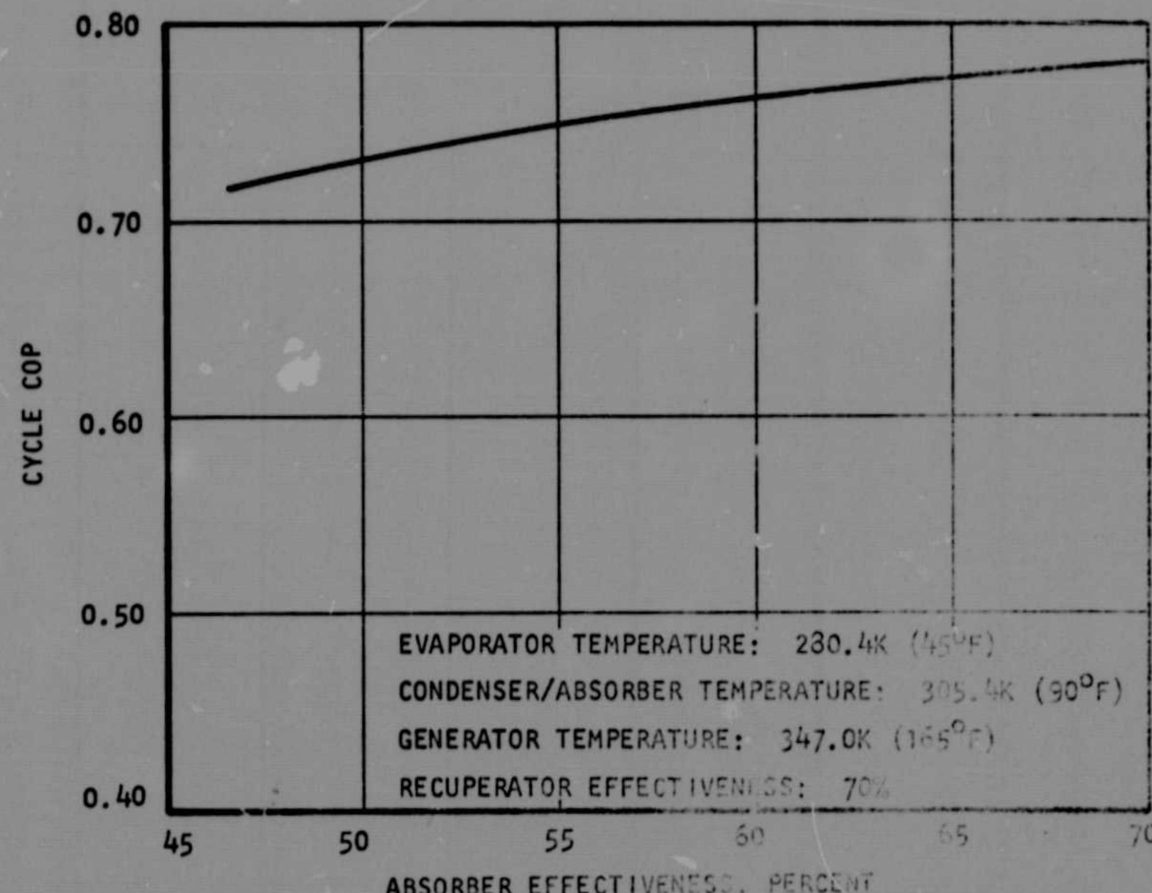


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Figure 2-4. Effect of Absorber and Recuperator Effectiveness on $LiBr/H_2O$ Refrigeration System Performance

SECTION 3

REVIEW OF RANKINE AIR CONDITIONER LITERATURE

A thorough review of Rankine-power systems and equipment has been published recently by Hittman Associates, Inc. (Reference 1-2.) This review was conducted under NSF Contract C858. Emphasis was placed on the Rankine power loop, and the air conditioner portion of the overall system was not presented at the same level of detail. Some of the information contained in Reference 1-2 is discussed below in the context of the current contract. Also, additional programs not covered in the Hittman survey are summarized. The investigations conducted by the following organizations currently active in this field are summarized:

(a) Barber-Nichols Engineering

(b) General Electric Co.

(c) AiResearch Manufacturing Company

(d) United Aircraft Research Laboratories

(e) Battelle Memorial Institute

(f) Thermo Electron Corporation

HITTMAN SURVEY DATA

The Rankine power system and equipment data compiled by Curran et al. are summarized in Table 3-1 (from Reference 1-2). Most of the Rankine engines listed were designed for electrical power generation from relatively high grade thermal energy; also, the power output of these machines far exceeds the requirements of a 10.5- to 17.6-kw (3- to 5-ton) refrigerant compressor (about 3 kw). However, the data are significant because the efficiencies attained with various types of expanders and liquid pumps are representative of existing design technology. Also, the machines listed are experimental units and in general have not been subjected to extensive developmental efforts aimed at maximizing efficiency. The data show that expander efficiencies between 70 and 80 percent can be achieved with various types of machines (turbine, rotary, and reciprocating) over a relatively wide range of power output.

Cycle efficiency, however, is dependent on the operating temperature levels of the boiler and condenser. Data from Reference 1-2 were used in the preparation of Figure 3-1. The overall efficiency shown is defined as follows.

$$\text{Overall cycle efficiency} = \frac{\text{net Rankine-cycle power}}{\text{boiler heat input}}$$

TABLE 3-1

SUMMARY OF CHARACTERISTICS FOR SMALL RANKINE ENGINES (FROM REFERENCE 1-2)

Manufacturer/ No. Designer	Type of Fluid	Type of Expander	Type of Application	Conceptual/ Experimental/ Production C/E/P	Cycle Conditions: Expander Inlet °F/psia	Cycle Conditions: Condenser °F/psia	Rated Power hp	Expander: Efficiency %	Expander: RPM @ Rated Power	Pump: Efficiency %	Pump: Power hp	Condenser Fan Power hp	Actual Overall Cycle Efficiency %	Estimated Service Life Years
1. Aerojet-Liquid Rocket	AEP-78	Turbine	Automobile	E	650/1000	201/35	74.9	69	26,500	60	25.6	9.7	10*	
2. Barber-Nichols	R113	Turbine	Solar Cooling	E	200/57	95/9.5	2.7	75	51,100	50	.1		8.3*	
3. Energy Research	Water	Rotary	Automobile	C	1200/3500	281/50	100.0	78		80			19.1	
4. Fairchild-Hiller	FC75	Turbine	Total Energy Plant	E	428/266	217/14.5	25.34	77.7	20,000				**	
5. Keller	Water	Rotary	Solar Power Plant	C	400/250	160/5	Any	80	1,800	80			16.	20
6. Keller	Water	Rotary	Automobile	C	550/1000	275/45	50-200	80	200-2000	80		0.5	18.	10
7. Kinetics	R113	Rotary	Automobile	E	375/355	200/55	47.0	70	3,500	85	4.0	5-7	***	
8. Kinetics	R114	Rotary	Solar Cooling	E	200/180	80/35	7.5	70	2,500		0.32		10.2	
9. Lear	Water	Turbine	Automobile	E	1100/1100	222/19	82.7	62	65,000		1.77	2.01	17.6	
10. Ormat	MCB	Turbine	Powerpack	P	Variable		3.0	70	20,000				6.0	20
11. Philco-Ford	MIPB	Turbine	Powerpack	E	600/37	374/1	0.13	54					5.0	
12. Steam Engine Systems	Water	Reciprocating	Automobile	E	550/750	228/20	135.0		2,750				**	
13. Scientific Energy Systems	Water	Reciprocating	Automobile	E	1000/1000	230/20	140.0	70	500-2000	80	20	7+	17.2	16-20
14. Sundstrand Aviation	CP-25	Turbine	Total Energy Plant	E	825/195	137/3.4	134.1	70	25,200		30-40		20.0	25
15. Sundstrand Aviation	Dowtherm A	Turbine	Powerpack	E	700/70	255/.5	8.05	72	24,000				13.4	20
16. Thermo Electron	Fluorinol 85	Reciprocating	Automobile	E	600/700	208/34	145.5	70	1,800	80	11.2	13.6	[illegible]	
17. Thermo Electron	Fluorinol 85	Turbine	Automobile	E	600/700	208/34	145.5	70	30,000	70	11.0	13.6	16.0	
18. Thermo Electron	CP-34	Reciprocating	Powerpack	E	550/500	220/25	5.0	75	3,600	60	0.37	0.25	13.0	
19. Thermo Electron	R114	Reciprocating	Solar Cooling	C	212/265	120/63	3.15	72	1,800		0.33		8.0	
20. United Aircraft	R113	Turbine	Solar Cooling	C	200-375/ 70-340	125/16.8	4.3	80	50,000				**	
21. United Aircraft	R114	Turbine	Solar Cooling	E	250-275/ 250-400	120/63	8.0	70	27,000				**	

* Not given in Reference. Calculated by HAI from data in Reference.

+ Value assumed for efficiency calculation

** Not given in Reference. Insufficient data for calculation.

*** Final results of current research not yet available.

where the net Rankine-cycle power is the expander power less the power necessary to drive the loop pumps, fans, and controls. The overall efficiencies of Figure 3-1 represent 55 to 65 percent of the Carnot efficiencies corresponding to the source/sink temperature shown, with the higher Carnot approach occurring at the lower boiler temperatures.

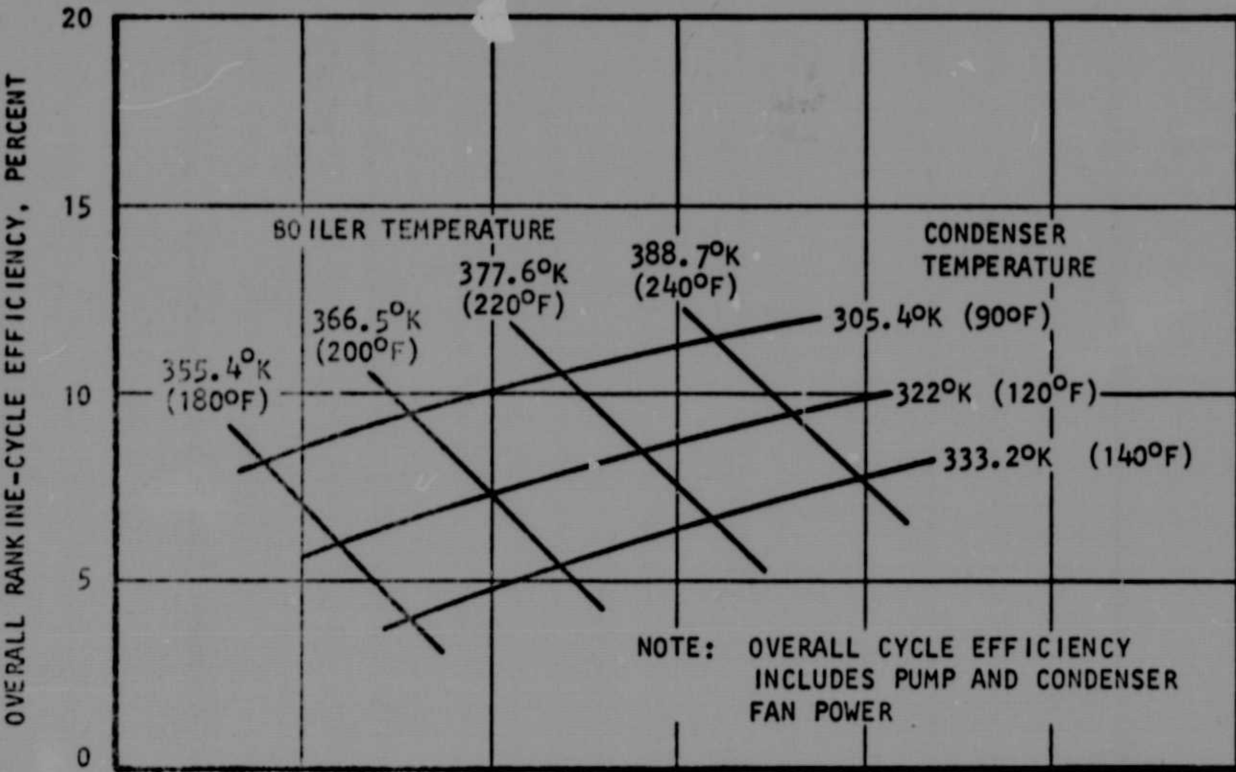


Figure 3-1. State-of-the-Art Rankine Engine Efficiencies

BARBER-NICHOLS INVESTIGATIONS (REFERENCES 3-1 AND 3-2)

Barber-Nichols has developed a solar-powered Rankine cycle air conditioner for installation and evaluation testing in the Honeywell transportable solar laboratory. This program was conducted under joint sponsorship of NSF (Grant PTP 74-01555) and Honeywell, Inc. A schematic of the solar-powered air conditioner is shown in Figure 3-2 (from Reference 3-2). A motor-generator is used to supplement the Rankine turbine when solar heat is not adequate to drive the air conditioning compressor. When the turbine output exceeds the requirements of the compressor, electric power can be generated. The system is designed to provide 3 tons of refrigeration with a collector water temperature of 374.8°K (215°F) and a condenser water temperature of 302.6°K (85°F).

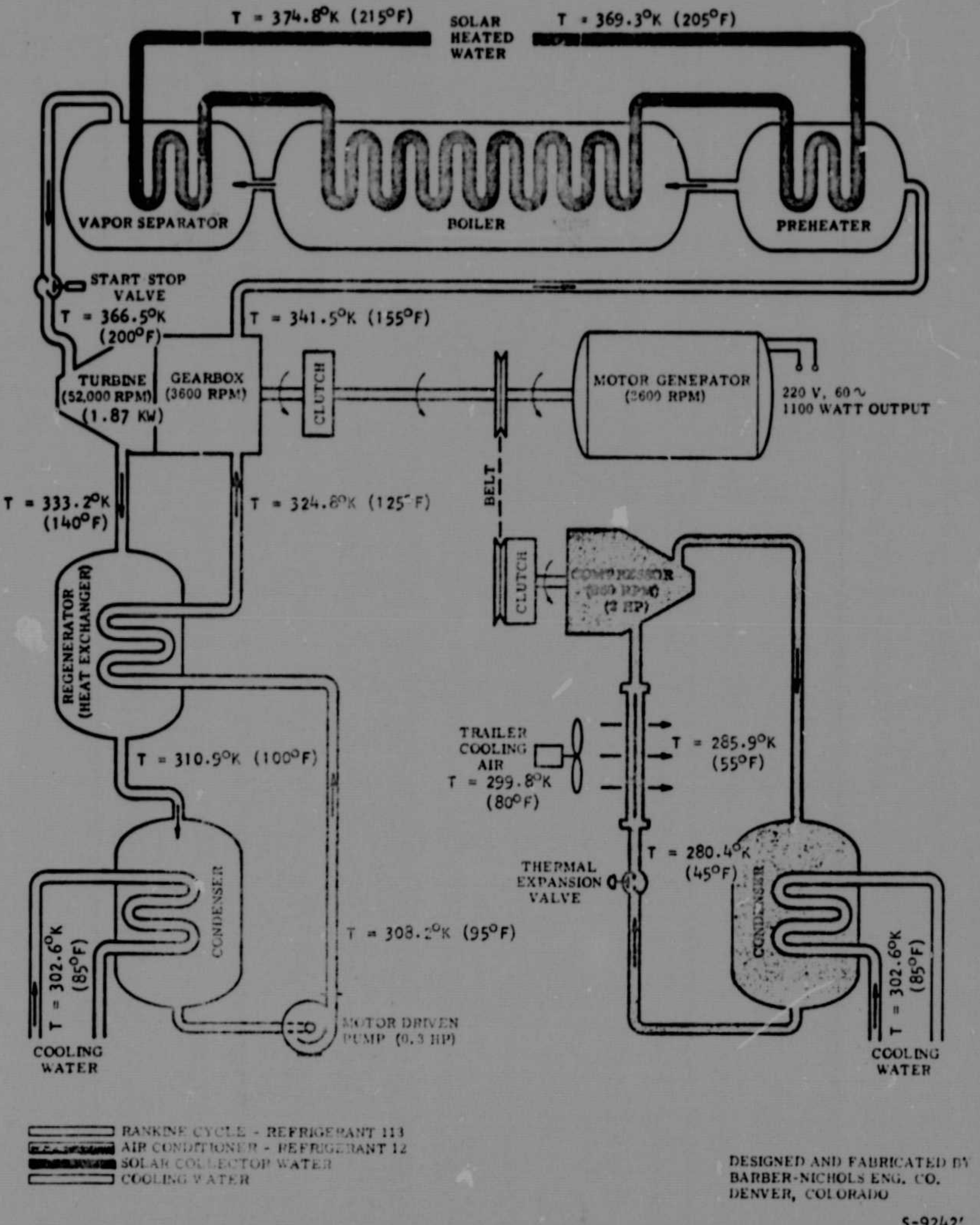


Figure 3-2. Barker-Nichols Solar-Powered Rankine-Cycle Air Conditioner (From Reference 3-2)

System overall performance (COP) is shown in Figure 3-3 as a function of collector water temperature. The data were obtained prior to installation in the solar laboratory; an overall COP of 0.5 was achieved. Through further development and relatively minor component and system improvements, it is anticipated that the overall COP could be increased significantly (above the 0.6 value at design point). It is significant that the system did operate with collector water temperatures as low as 349.8°K (170°F). This is 25°K (45°F) below design value. Under this condition system capacity drops from 10.5 to 3.5 kw (3 to 1 ton); also, the COP decreases from 0.5 to about 0.25. Data presented in Reference 3-1 show that a turbine efficiency of 72 percent was achieved; also, the efficiency could be maintained above 65 percent by maintaining turbine speed at design value through the use of the electric motor.

GENERAL ELECTRIC

General Electric Space Division has been engaged in the development of a multi-vane expander for low-temperature Rankine cycle application for about 4 years. A 4.85-kw Freon unit is currently in test. Expander efficiencies as high as 75 percent have been achieved, and through development it is anticipated that this value can be increased to 85 percent. Figure 3-4 (from Reference 3-3) shows the expander configuration and its performance over a range of flow. A 10.5-kw (3-ton) commercial refrigerant compressor was coupled to a multivane expander and tested over a range of temperatures. The unit has been subjected to 1020 hr of unattended endurance testing. A COP of about 0.5 was achieved, corresponding to the following conditions

Expander efficiency = 72 percent

Compressor efficiency = 60 percent

Condenser temperature = 310.9°K (100°F)

Expander inlet temperature = 366.5°K (200°F)

The major advantage of a vane expander is the high torque at low speed.

AIRESEARCH INVESTIGATIONS

In 1970 AiResearch delivered two heat-powered refrigeration systems to the U.S. Army (MERDC). Results from this development program are presented in Reference 3-4. One of these systems was an air conditioner and the other a water chiller of similar design. These systems utilized the thermal energy contained in the exhaust stream of a gas turbine to produce mechanical energy through the Rankine cycle. This energy is expended in driving (directly) the centrifugal compressor of a refrigeration loop. A schematic of the system is shown in Figure 3-5. The turbocompressor is a hermetic unit featuring a two-stage compressor driven by a single-stage turbine at about 48,000 rpm. A photograph of the unit is shown in Figure 3-6. Compressor pressure ratio at design point was 6.2:1. Turbine and compressor efficiencies of 80 and 75 percent, respectively, were consistently measured in tests. The system

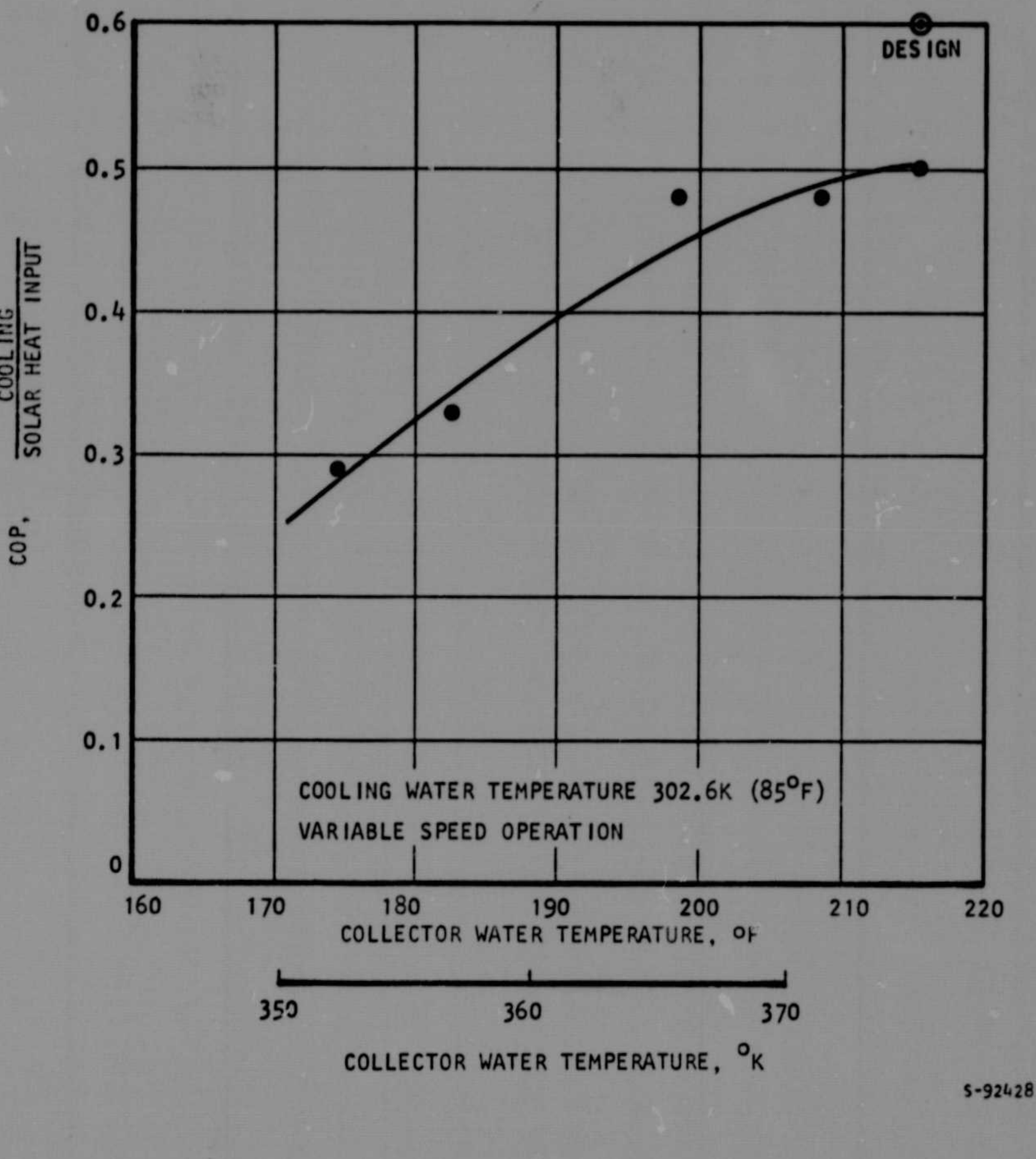
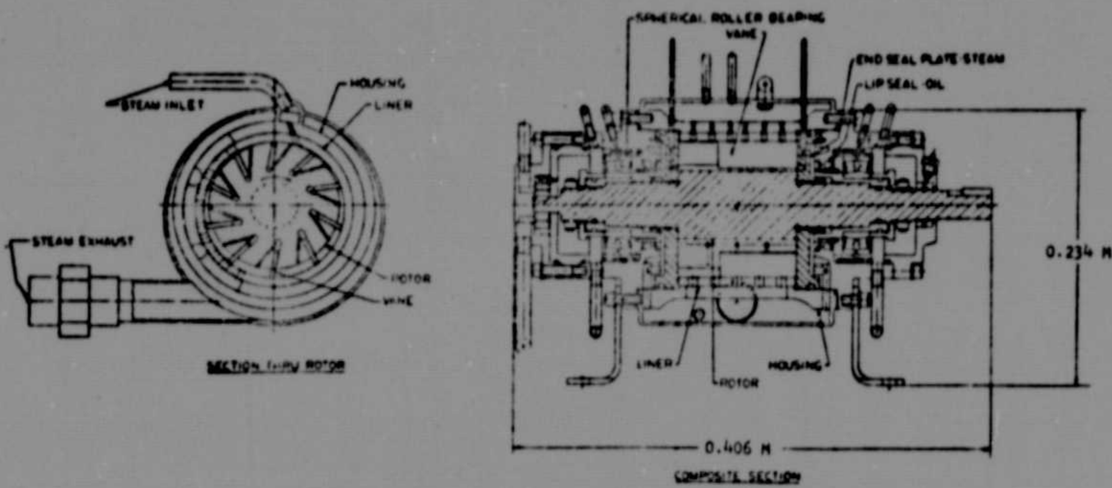
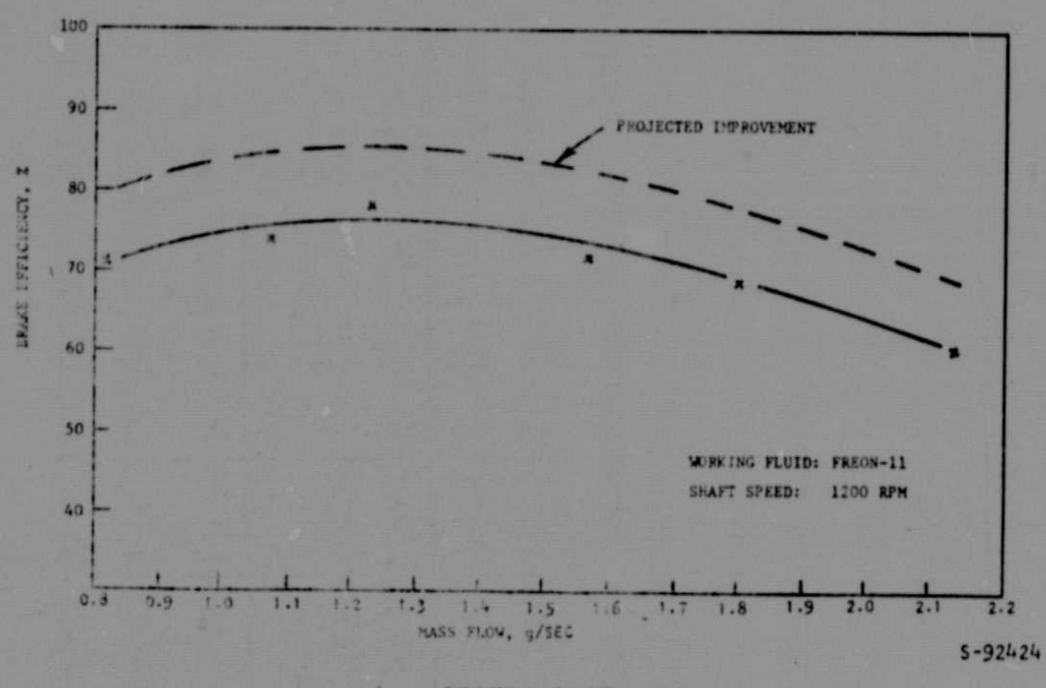


Figure 3-3. Barber Nichols System Test Performance (from Reference 3-2)



a. CONFIGURATION



b. PERFORMANCE

Figure 3-4. General Electric Multi-Vane Expander
(From Reference 3-3)

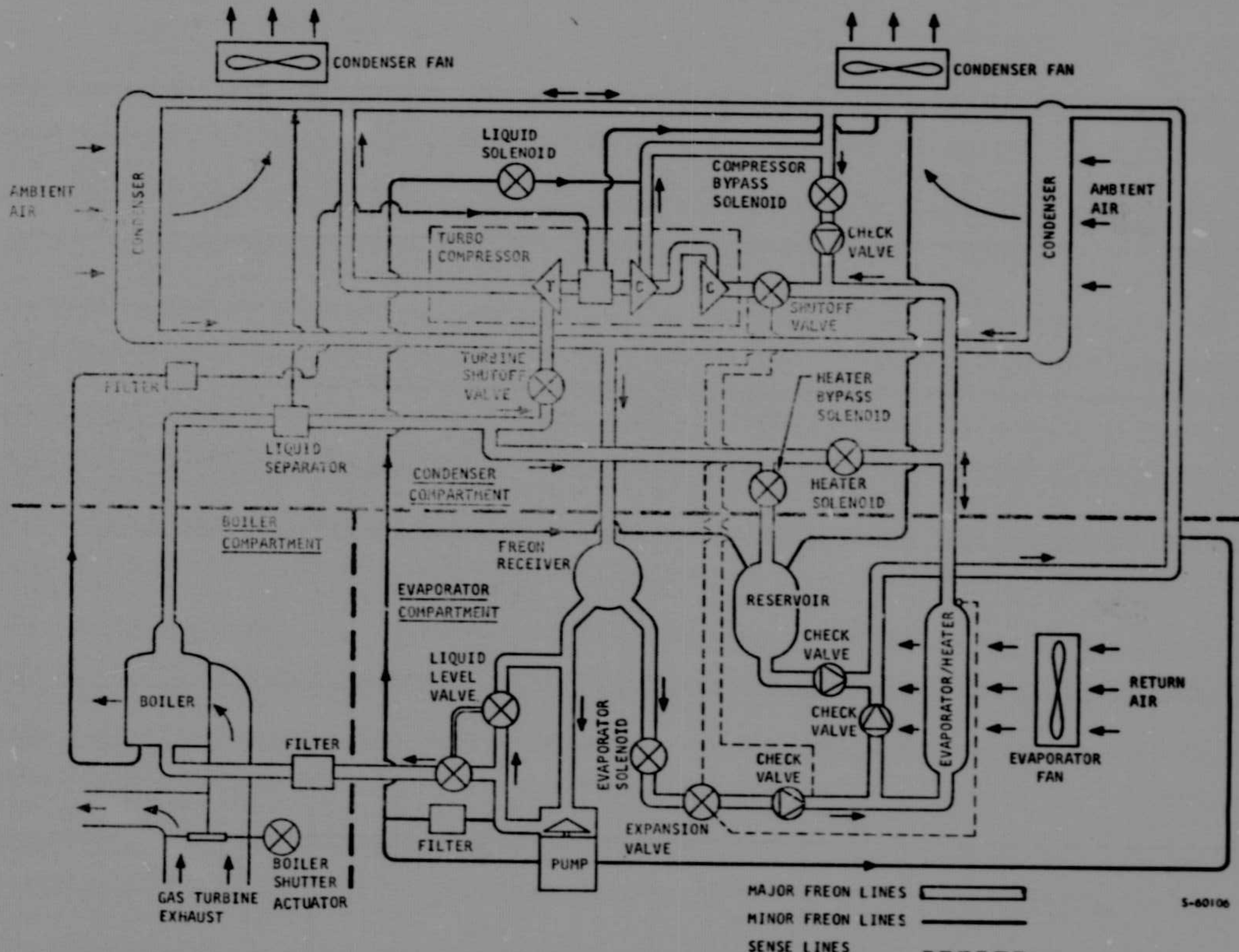


Figure 3-5. Waste Heat Rankine Refrigeration Unit (Detailed Schematic)

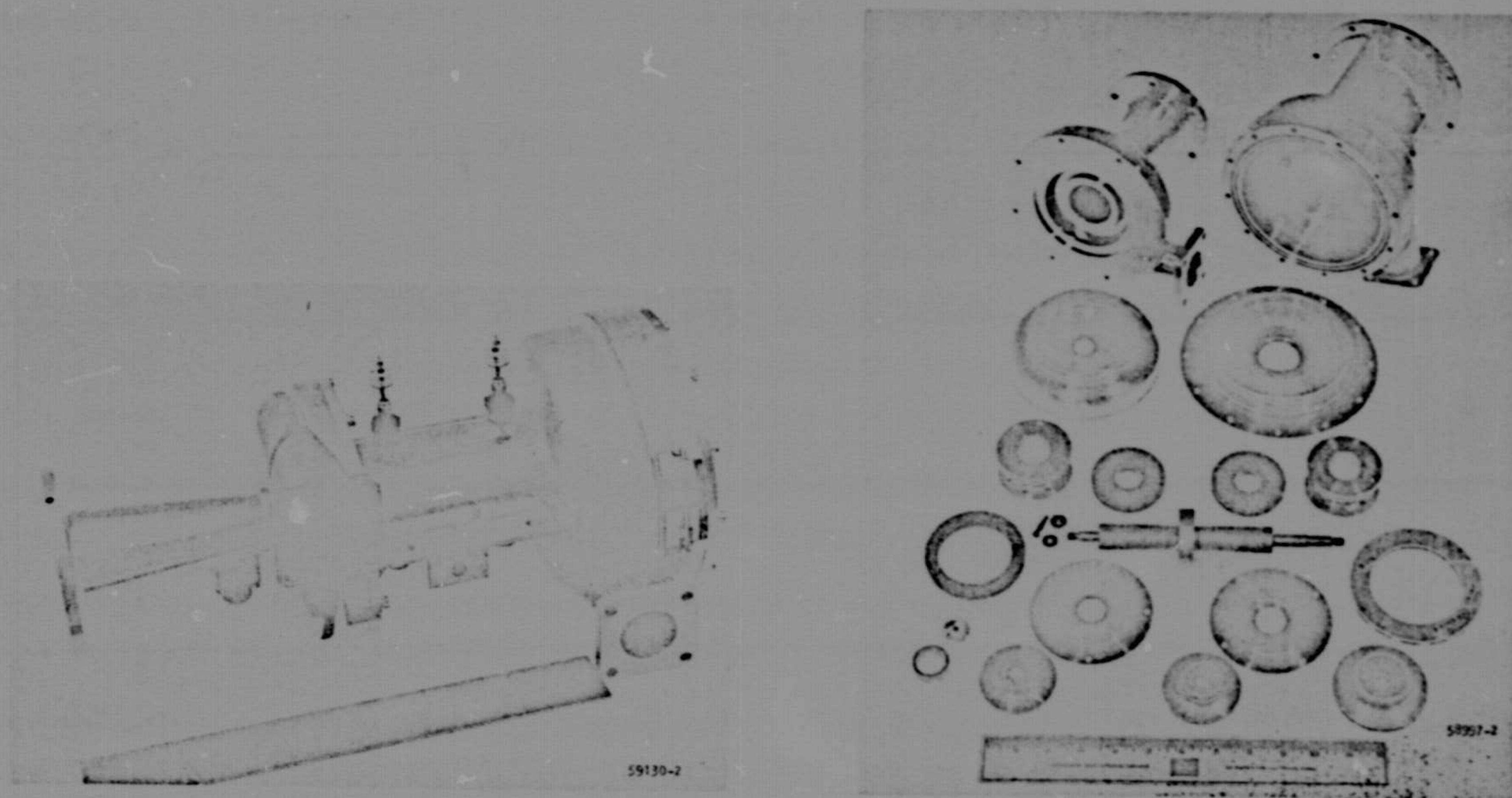


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Figure 3-6. Waste Heat System Turbocompressor

performance as obtained in testing and the estimated performance of the system slightly modified for operation from a solar heat source are shown in Table 3-2.

Currently, AiResearch is engaged in the development of a 35.2-kw (10-ton) R-12 centrifugal compressor. The test unit shown in Figure 3-7 is driven through a magnetic coupling so as to permit the use of static seals throughout. The 0.038-m (1.5-in.) dia compressor wheel is designed to rotate at 90,000 rpm. Efficiencies as high as 75 percent are anticipated from this machine operating at the conditions listed below.

Inlet pressure: 386 kN/m^2 (56 psia)

Inlet temperature: 286.1°K (55°F)

Outlet pressure: 1275.5 kN/m^2 (185 psia)

R-12 flow rate: 0.302 kg/sec (40 lb/min)

Speed: 90,000 rpm

Test data obtained to date have indicated very close agreement between design and experimental values.

An investigation was made of the practicality of using high-speed turbomachinery in systems designed for residential applications. To this end, data on high-speed rotating equipment was obtained from the AiResearch Industrial Division (AID) of The Garrett Corporation. AID is the worlds leading manufacturer of turbochargers for internal combustion engines, with a total production of 350,000 units per year that represents a family of several models. Some of these units are shown in Figure 3-8, and Figure 3-9 lists the speed, size, and efficiencies of these machines. In a turbocharging application, the maximum pressure ratios ever necessary are less than 3:1, which is representative of the capability of these machines. The turbine efficiencies listed include the mechanical losses of the machine and represent the net shaft power delivered to the compressor. The size and construction of these machines are not representative of a low-temperature turbocompressor, as would be required for a solar-powered air conditioner. For example, the turbine wheel and casing are fabricated of high-temperature alloys; also, the volumetric flow rates through the unit are generally much larger than would be required in a refrigeration turbocompressor. However, the data indicate that manufacturing technology is available for large production of turbomachines of relatively sophisticated aerodynamic design. Projected factory cost for production quantities on the order of 1/2 million units per year for 5 years is estimated at $60 per unit. Smaller size and lower temperature units could be fabricated at much lower cost. Manufacturing techniques using glass-reinforced polycarbonate could possibly be used to advantage.

TABLE 3-2

PERFORMANCE OF WASTE HEAT REFRIGERATOR MODIFIED FOR SOLAR OPERATION

	Current Design	Solar Modification
Refrigerant	R-11	R-11
Waste heat source	Gas at 672.0°K (750°F)	Water at 366.5°K (200°F)
Refrigerated air		
Flow	1.03 m^3/sec (2200 cfm)	1.23 m^3/sec (2600 cfm)
Return temperature	305.4°K (90°F)	299.8°K (80°F)
Outlet temperature	291.5°K (65°F)	288.7°K (60°F)
Cooling air		
Flow	4.48 m^3/sec (9500 cfm)	4.48 m^3/sec (9500 cfm)
Inlet temperature	322.0°K (120°F)	308.2°K (95°F)
Boiling temperature	410.9°K (280°F)	358.2°K (185°F)
Condensing temperature	335.9°K (145°F)	318.7°K (114°F)
Evaporating temperature	283.2°K (50°F)	283.2°K (50°F)
Turbine pressure ratio	5.1:1	3.0:1
Compressor pressure ratio	6.2:1	3.4:1
Refrigeration	17.6kw (5 tons)	15.8kw (4.5 tons)

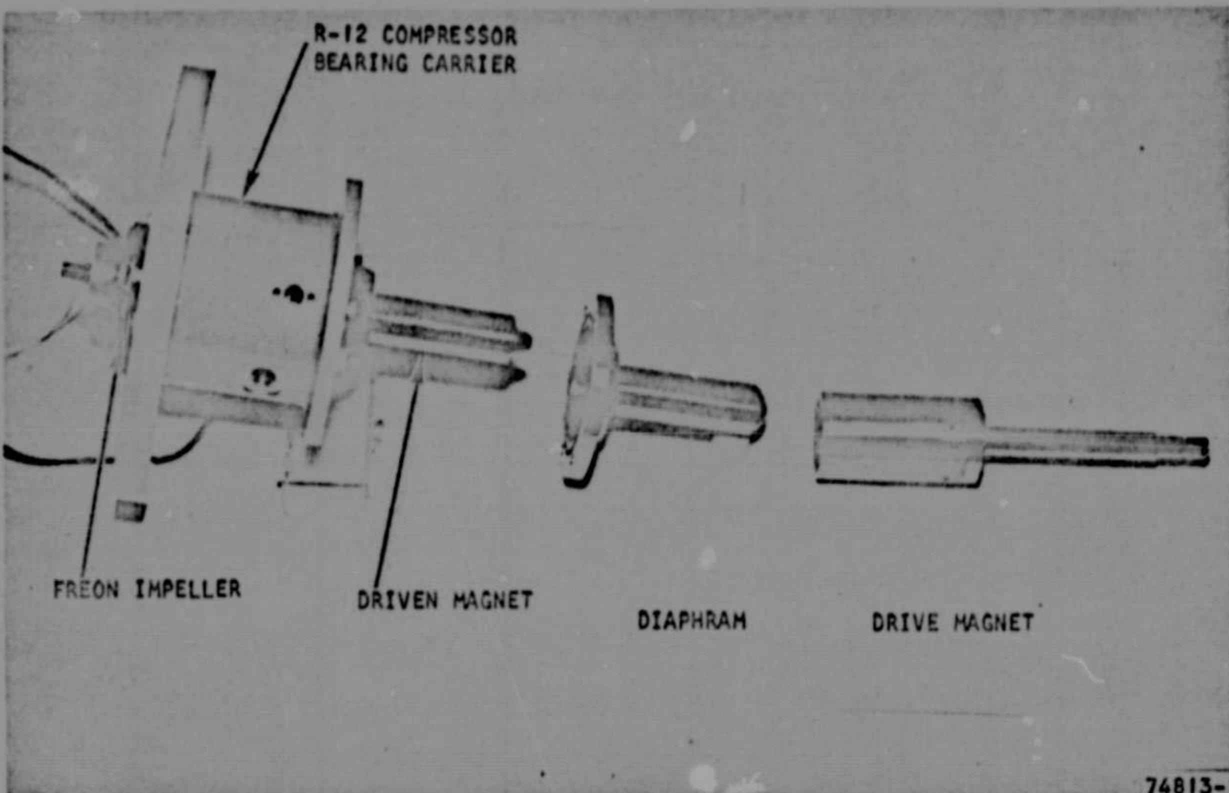

Figure 3-7. 35.2-kw (10-Ton) Prototype Freon Compressor

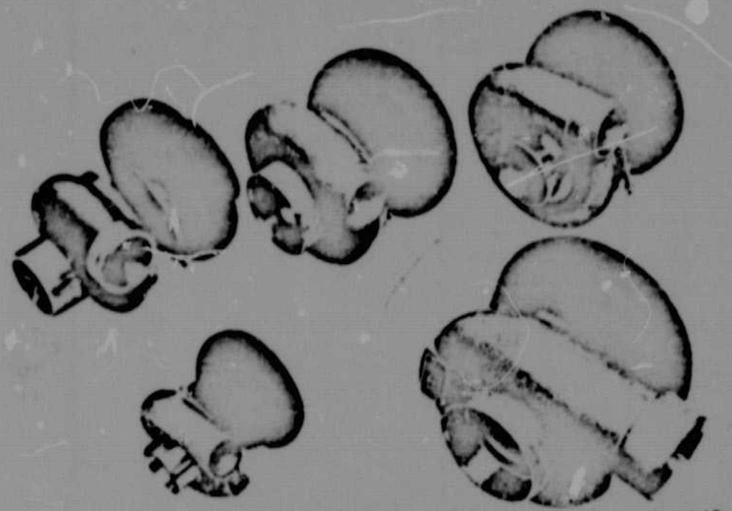

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Figure 3-8. Automotive Turbochargers

AID TURBOCHARGER DATA

Model No.	Speed, rpm	Turbine Diameter, in.	Turbine Efficiency, percent	Compressor Diameter, in.	Compressor Efficiency, percent
T04/T04B	133,000	2.92	74	2.75	74
TE06/T12	113,000	3.5	71	3.3	72
TV61	113,000	3.5	73	3.4	74
TV71	106,000	3.8	73	3.7	74
T18A	83,000	5.1	75	5.0	75

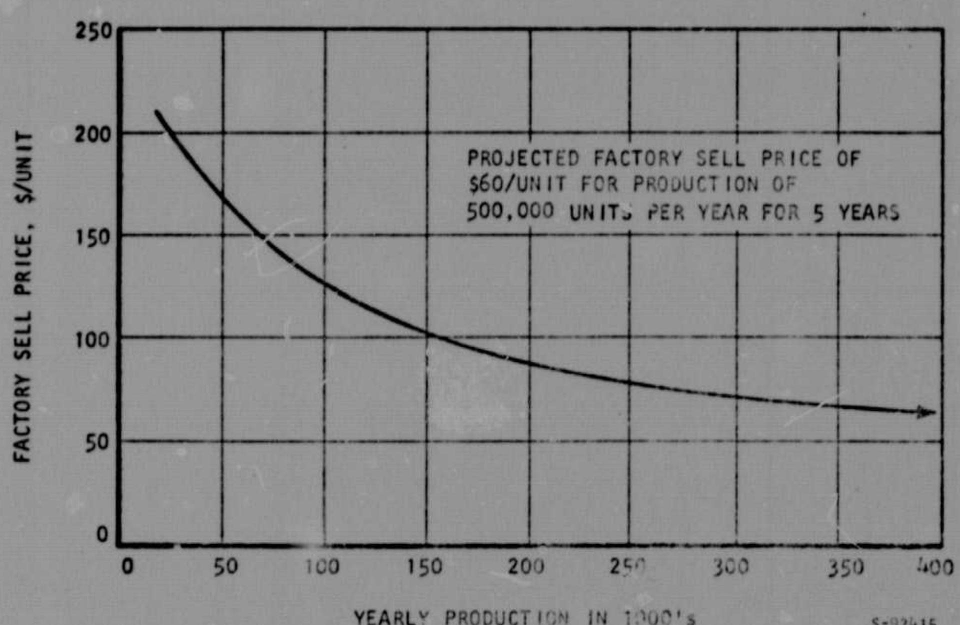

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Figure 3-9. Turbocharger Performance and Cost Data

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UNITED AIRCRAFT RESEARCH LABORATORIES (REFERENCES 3-5 AND 3-6)

United Aircraft is currently engaged in the development of a turbocompressor applicable to a solar heat-powered Rankine air conditioner. The objectives of this program are to (1) demonstrate the feasibility of operating a turbocompressor air conditioner powered by heat collected from a state-of-the-art solar collector and (2) analytically demonstrate the potential of such a system. The demonstration system will utilize a modified existing turbocompressor unit originally designed to produce 28.1 kw (8 tons) of refrigeration with R-114 as the working fluid and gas turbine exhaust as the heat source. Figure 3-10 (from Reference 3-6) shows the unit, which consists of a two-stage compressor driven by a single-stage turbine. The rotating assembly is supported by ball bearings; rotational speed is about 30,000 rpm. Peak efficiencies for the compressor and turbine operating as separate components have been demonstrated in tests to be 0.69 and 0.78, respectively.

Matching of the modified compressor and turbine at the air conditioner design point will result in considerable turbine efficiency drop. Compressor and turbine efficiencies of 68 and 66 percent, respectively, are predicted (Reference 3-3) under the following system conditions.

Working fluid: R-11

Turbine inlet temperature: 366.5°K (200°F)

Condensing temperature: 313.7°K (105°F)

Evaporating temperature: 280.4°K (45°F)

The corresponding overall system COP (evaporator load/boiler heat input) is estimated at 0.366.

Investigations pursued by Hamilton Standard (a division of United Aircraft) indicate that compressor efficiencies as high as 80 percent could be achieved in the size necessary to produce 2 to 5 tons of refrigeration. This estimate is supported by test data obtained on an advanced compressor design fabricated from sheet stock. Similarly, with limited development work turbine efficiencies of 80 percent could realistically be obtained. With these efficiencies, a system COP of 0.76 can be achieved at the baseline conditions listed above.

BATTELLE MEMORIAL INSTITUTE (REFERENCE 3-7)

Battelle is currently under contract to NSF for the design study of a heat pump using a rotary vane compressor-expander. This rotary machine features pivoting vanes hydrodynamically lubricated by the working fluid. The basic design of the unit was derived from a high-speed aircraft hydraulic pump developed for the Air Force. It is anticipated that the pivoting vane design

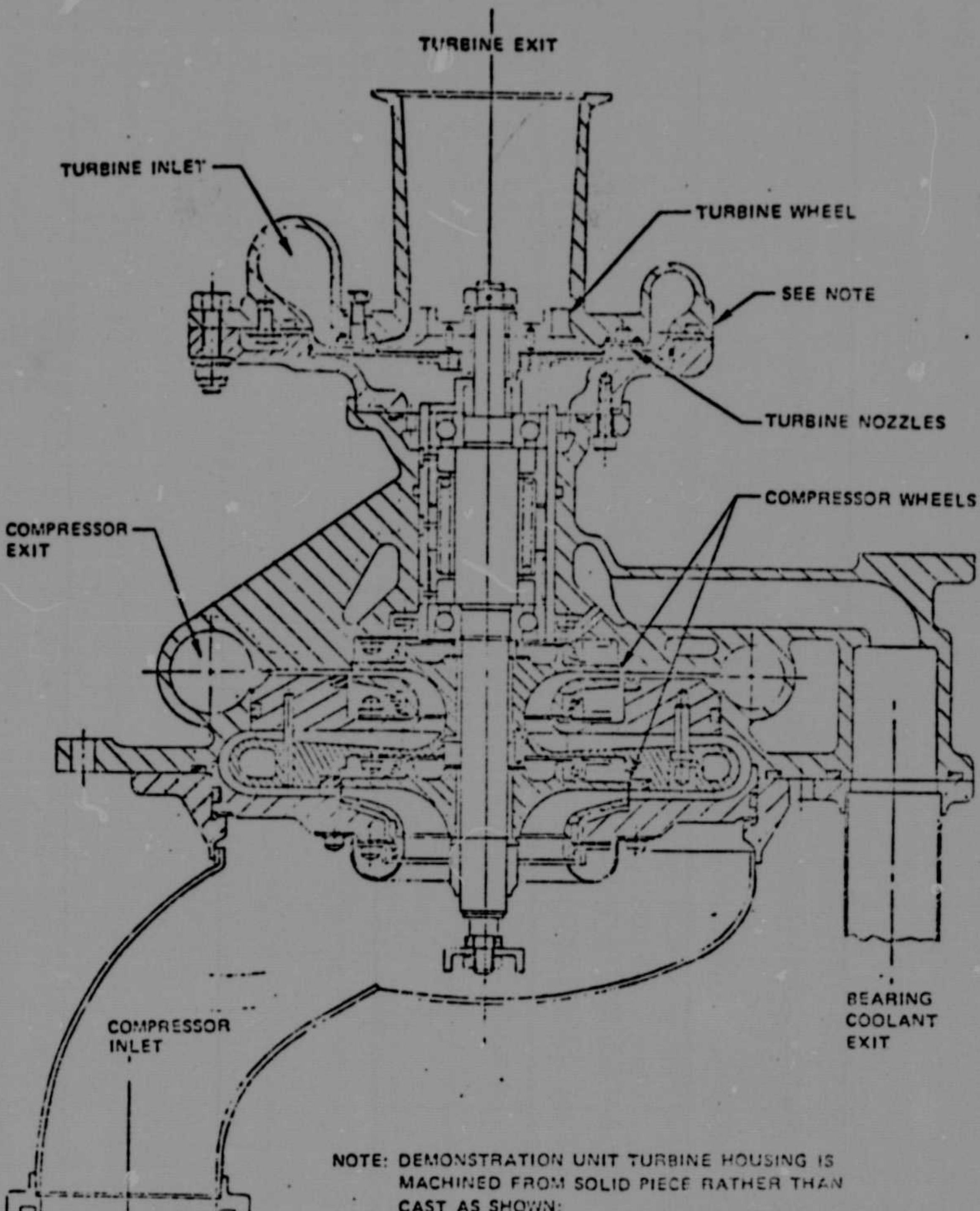

Figure 3-10. UARL Turbocompressor Test Unit
(From Reference 3-6)

will provide very high compressor and expander efficiencies at low rotational speed and low cost. However, only qualitative data on the machine are reported in Reference 3-7.

THERMO ELECTRON CORPORATION (REFERENCES 3-8 AND 3-9)

Thermo Electron Corporation has been engaged in the development of Rankine power systems using reciprocating machines since 1963. Currently, this organization is involved in the development of a single-cylinder 5-hp engine model. The engine is designed to operate with CP-34 as the working fluid. Design data are listed below.

Inlet pressure: 3446 kN/m^2 (500 psia)

Inlet temperature: 560.9°K (550°F)

Release pressure: 172 kN/m^2 (25 psia)

Engine rpm: 3600

Engine bore: 0.057 m (2-1/4 in.)

Engine stroke: 0.044 m (1-3/4 in.)

No actual test data on this engine are presented in Reference 3-8. However, the characteristics for a 5-hp portable package using this basic engine are reported in Table 3-3 (from Reference 3-8).

Currently, Thermo Electron Corporation is engaged in the development of a gas-fired heating/cooling system using the reciprocating engine Rankine-power system principle. The working fluid is R-22 in both the Rankine power loop and the cooling loop. Estimated characteristics of this machine are listed in Table 3-4 (taken from Reference 3-8). The gas COP is defined as the ratio of net cooling rate to thermal input to the boiler during cooling.

Using basic reciprocating expander technology, the solar-powered heating/cooling system depicted in Figure 3-11 (from Reference 3-9) is suggested by Thermo Electron Corporation. Expander, compressor, and feed pump efficiencies of 72, 72, and 80 percent, respectively, are claimed as consistent with efficiencies measured in tests of similar equipment. With R-114 as the power loop working fluid and R-22 as the refrigerant, an overall COP (cooling effect/heat input) of 0.6 is predicted; this corresponds to a boiler temperature of 377.6°K (220°F) and a condensing temperature of 313.7°K (105°F).

TABLE 3-3

SYSTEM CHARACTERISTICS FOR 5-HP
PORTABLE POWER PACKAGE
(FROM REFERENCE 3-8)

Working fluid	Cp-34
Flow rate	392 lb/hr
Net engine shaft power	12,700 Btu/hr
	5 hp
Feed pump power	950 Btu/hr
Accessories drive power	1,600 Btu/hr
Total engine shaft power	15,250 Btu/hr
	6.0 hp
Q boiler	80,400 Btu/hr
Q fuel	98,100 Btu/hr
Q regenerator	13,800 Btu/hr
Q condenser	64,200 Btu/hr
η overall	13.0 percent
η cycle (does not include boiler efficiency of 82 percent)	15.8 percent
Engine overall efficiency	75 percent
Feed pump overall efficiency	60 percent
Regenerator effectiveness	90 percent
Boiler efficiency	82 percent

TABLE 3-4

CHARACTERISTICS OF COST-OPTIMIZED
HEATING/COOLING SYSTEM
(FROM REFERENCE 3-8)

Cooling rate	36,000 Btu/hr
Heating rate (input)	122,000 Btu/hr
Gas COP net	0.4
Weight (condensing section)	410 lb
Size (condensing section)	4 by 3 by 2.3 ft
Electric requirements	
Heating	0.3 kw
Cooling	0.5 kw
Above values based on:	
Engine overall efficiency	70 percent
Compressor overall efficiency	70 percent
Vapor generator thermal efficiency	82 percent
Feed pump overall efficiency	60 percent
Evaporator temperature	45°F
Ambient air temperature	95°F
Boiler outlet temperature	405°F

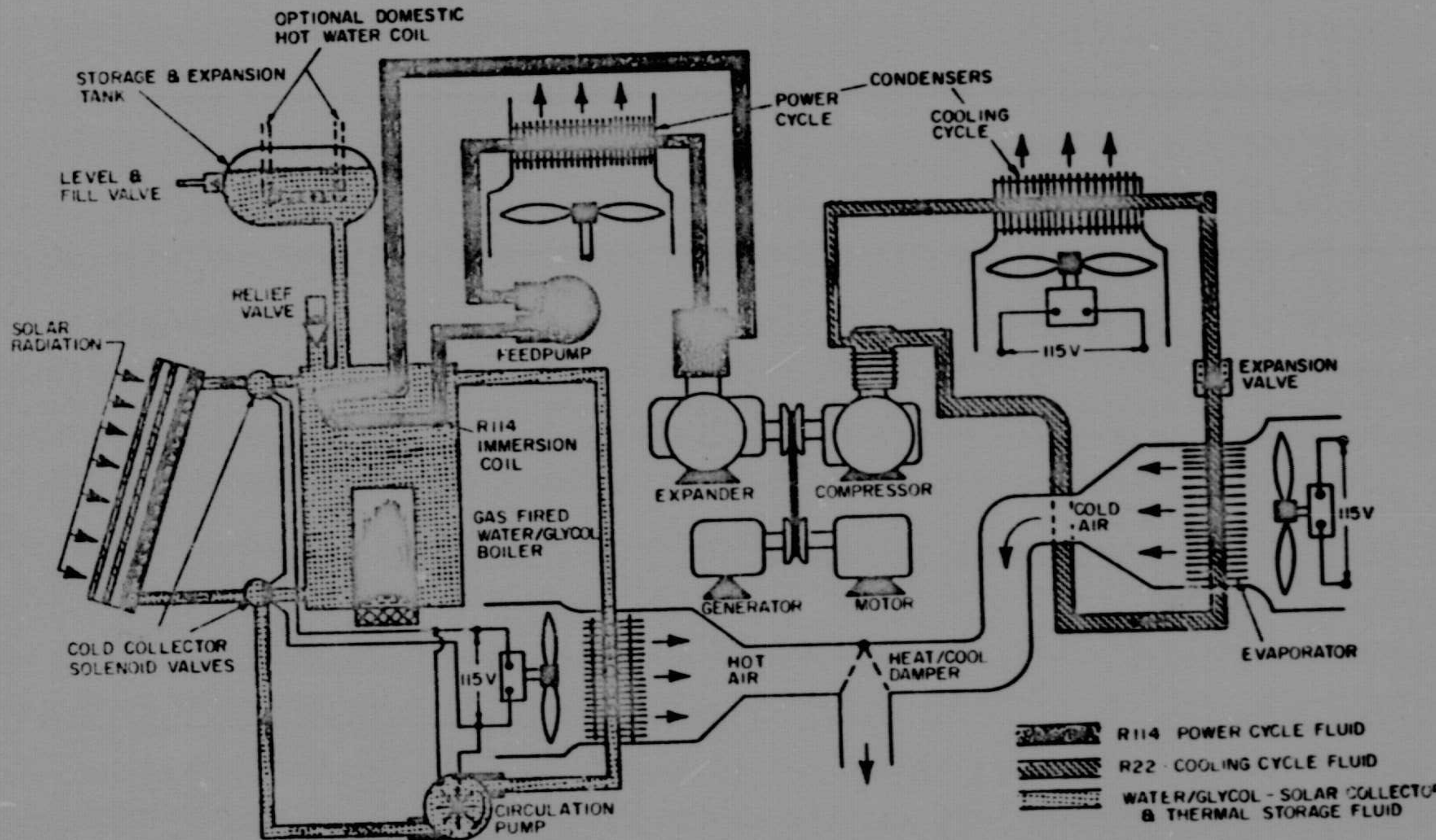

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Figure 3-11. Schematic of a Rankine-Cycle Driven Combined Cooling/Heating System Using a Solar Energy Heat Source (From Reference 3-9)

SECTION 4

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